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Interim Report

CENTAUR STRUCTURAL CAPABILITY STUDY

Contract No. NAS 3-3228
Task Order No. 28

Prepared for the
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National Aeronautics and Space Administration
Cleveland, Ohio

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REVISIONS

NO.	DATE	BY	CHANGE	PAGES AFFECTED
	6-19-5	EEM	Preliminary Outline and List of Ref.	11
	8-19-5	EEM	Para.1.4.2-"Probabilistic Considerations for Allowables, Three Sigma (3σ) Loads and Factors of Safety."	52 + v
	9-19-5	EEM	Para.1.3.2.3.1,"Local Loading Effects on Tank Skins".	50 + v
	10-19-5	EEM	Para. 1.3.4.,"Buckling Criteria".	77 + vi
	12-19-5	WRL	Para. 1.2, "Structural Description (and Capability)-Centaur".	83 + xii
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	2-21-66	EEM	Para.1.4.1,"Fatigue Spectra, Allowables and Analysis."	22 + iv

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SUMMARY

This interim report constitutes a portion of the monthly progress report submitted under Contract No. NAS 3-3228, Task Order No. 28, "Centaur Structural Capability Study," The primary content of this interim report consists of Section 1.4.1, "Fatigue Spectra, Allowables and Analysis." This section is similarly numbered in the "Preliminary Outline of the Final Report," previously submitted. A list of references for this section is also included. Some of these references differ, in whole or in part, from those shown in the previously submitted, "Preliminary List of References for Task I".

1.4.1 Fatigue Spectra, Allowables and Analysis

All machines or structures must be designed to satisfy three basic requirements: they must perform their intended function, they must have adequate service life, and they must be capable of being produced, sold and maintained at reasonable cost. Fatigue can seriously impair the fulfillment of all of these requirements. Fatigue strength becomes increasingly important in aerospace vehicle structural design; as excess weight (hidden margins) and factors of safety are reduced; when mean flight loads approach maximum expected flight loads (small deviation about a high mean); whenever higher ultimate tensile strength alloys are employed to increase static strength (and reduce weight); and whenever structures are exposed to low temperature or vibratory environments.

For adequate structural design of advanced missiles, aerospace vehicles and recoverable boosters, the effects of fatigue on vehicle service life and the consequences of catastrophic fatigue failure must be determined. Requirements for fatigue stress analysis are becoming increasingly prevalent in requests for bids on vehicles issued by contracting agencies. Careful consideration of fatigue during the study stage and throughout the design phase will help ensure efficient designs. Constant advanced development in the field of structural fatigue is an astronautical engineering prime requirement.

Any rational investigation of the fatigue life of a structure involves: (1) an assumed or imposed fatigue loading spectrum, (2) an estimated or experimentally determined and evaluated fatigue design allowable strength level, and (3) an estimated or calculated analysis of the resulting safety (from failure caused by fatigue effects) of the structure.

The following sub-sections present the current design philosophy, utilized to give an estimated adequate fatigue life for Centaur, and suggest a rationale leading to alternative procedures, which could materially improve the accuracy of the investigation and, quite likely, reduce vehicle weight.

1.4.1.1 Current Practice - The following sub-sections present a summary of current practice used in the design analysis of Centaur to help obviate against fatigue problems. Primary emphasis is devoted to the treatment of propellant tank walls, since they constitute the largest portion of the primary structural weight (affected by fatigue considerations).

1.4.1.1.1 Criteria - The following is extracted from the Centaur (AC-6 to AC-15) Structural Design Criteria (Ref. M-1, Para. 2.9), and presents general fatigue considerations recommended as guidelines for Centaur design.

> "The design of the Centaur upper stage vehicle, and its components, shall incorporate those practices necessary for best fatigue life in every case where repeated or reversed stresses occur. Special attention shall be given vessels subjected to repeated pressurization and/or cryogenic cycles and structure adjacent to significant vibration sources. Where no conflict with other criteria is involved, materials with good fatigue behavior shall be used in such applications. Care shall be exercised to minimize detrimental residual stresses, stress concentrations, and poor surface finish. Where these cannot be avoided, consideration shall be given to their effect on fatigue life in the stress analysis of the pertinent items.

Telemetry packages and other electronic packages, which are subjected to maintenance pressurization cycles and flight differential pressures, must be designed and tested to pressure fatigue requirements considering a minimum of ten times the number of operating pressure cycles. Where specific service requirements are accurately known and a high degree of quality control is assured, a minimum of four times the number of operating pressure cycles shall be used. When service requirements are questionable, the prime design group shall suitably determine the number of expected service cycles. Proof pressure shall be 1.33 times the maximum positive and/or negative pressure expected in service."

1.4.1.1.2 Material/Joint Environmental Test Specification - The 200/300 cycle (no leakage at 200 ~ , no failure at 300 ~) material/joint acceptance criteria for hydrogen tank skins is presented in specification 0-71016 (Ref. M-2). The following pages present the body of the specification and indicate changes made in the original version (4-62) in revisions A (4-1-63) and B (9-20-63).

SPECIFICATION 0-71016

ENVIRONMENTAL TESTS

PROCEDURE FOR HYDROGEN TANK SKINS

1.0 Scope

This specification outlines the test procedure necessary to evaluate

Specification 0-71016 (Continued)

the joint fatigue properties at liquid hydrogen temperatures in order to guarantee material compliance with missile tank requirements.

2.0 Applicability

Requirements for testing per this specification will be made by a note on the Engineering drawing. The material must comply with paragraph 8.0 of this specification before major assembly of tank. Identification of part must be made as per paragraph 3.1.3 (k) of 0-71015 (Ref. M-3).

3.0 Sampling

At least one specimen shall be taken immediately adjacent to the skin and gore set for each coil and for each ship. When the length of cut material exceeds 80 feet, two specimens are required; one adjacent to each end of the cut length.

4.0 Specimen Configuration

The specimen configuration depends on the type of material, joint design, gage and orientation of material grain direction. Therefore, it will be necessary to obtain the proper test specimen drawing from the Stress Group for each part requiring tests per this specification.

5.0 Test Conditions

Test Temperatures:	-423° F
Cycle Rate:	6 cycles per minute

Specification 0-71016 (Continued)

Leak Check:

Material	Number Of Leak Checks	Number Of Cycles Before Leak Check			Maximum Number Of Cycles		
		4/62	4/63	9/63	4/62	4/63	9/63
GDC 0-71004(Ref. M-4)	1	200	200	200	500	--	--
GDC 0-71005(Ref. M-5)	1	200	200	200	500	--	--
GDC 0-71012(Ref. M-6)	1	75	200	200	300	--	--
GDC 0-71022(Ref. M-7)	1	--	200	200	--	--	--

6.0 Stress Level

The following stress levels are to be applied:

Material	4/62	4/63	9/63
GDC 0-71004	0-140 ksi	0-146 ksi	0-135 ksi
GDC 0-71005	0-120 ksi	0-110 ksi	0-110 ksi
GDC 0-71012	0-110 ksi	0- 95 ksi	0- 95 ksi
GDC 0-71022	------	0-146 ksi	0-135 ksi

7.0 Test Procedure

Cycle at temperature to number of cycles required for leak check.

Use dye check method to determine leak. After required leak checks are made, cycle specimens to failure or maximum number of cycles specified in 5.0. Stress levels are based on ram load divided by minimum skin cross section of each specimen. Cross section is obtained by a measurement to four places of specimen width and thickness.

8.0 Acceptance Levels

The following acceptance levels are to be met:

Specification 0-71016 (Continued)

Material	Cycles to Leak			Cycles to Fail		
	4/62	4/63	9/63	4/62	4/63	9/63
GDC 0-71004	200	200	200	300	300	300
GDC 0-71005	200	200	200	300	300	300
GDC 0-71012	75	200	200	100	300	300
GDC 0-71022	---	200	200	---	300	300

1.4.1.1.3 Design Application - The present Centaur tank fatigue cycle/membrane stress philosophy originated around the early (1962-1963) decision to increase tank wall thickness on the first vehicles in order to reduce any possible cracking and/or leaking problems (during normal service life) caused by poor spotwelds. The vehicle structural design criteria discusses general fatigue considerations. The environmental test specification (for joints) is utilized as a "go"-"no go" criterion for material acceptance. The specified uniaxial cyclic stress level from this test specification is employed in the regular stress analysis as the membrane allowable material yield stress. The combination of the (1) general criteria, (2) material environmental test specification and (3) use of the test stress level as an allowable in the stress analysis (for maximum static loads) results in an engineering device to provide some protection against fatigue problems. In addition, it saves the cost of a fatigue stress analysis; however, it does tend to increase vehicle structural weight and reduce analytical load capability.

1.4.1.2 Proposed Rationale - The following sub-sections present a summary of a rational foundation suggested for use in design fatigue stress analysis of Centaur tank skins. Again, primary emphasis is devoted to the treatment of propellant tank walls, since they constitute the largest portion of the primary structural weight (affected by fatigue considerations). In order to reduce vehicle weight, increase payload capability in this technical area, and provide adequate fatigue life, four requirements need to be satisfied. They are as follows: (1) develop a representative design fatigue loading spectrum, (2) develop or employ more realistic design allowable fatigue strength levels, (3) perform a fatigue stress analysis of the primary, structural elements (e.q., propellant tank walls) which could be fatigue problem areas, and (4) revert to the use of static material and joint strengths (at the appropriate temperatures) for providing the reaction to design maximum loads in the regular vehicle stress analysis.

1.4.1.2.1 Fatigue Spectra - Structural Design Fatigue Loading Spectra should be developed for a typical AC-6/AC-15 Centaur vehicle subjected to heavy service conditions. The load-time history covered should extend from fabrication through completion of the flight mission and be presented in a manner to show cyclic variations in stress (or force) in the major structural components. Typical loading conditions might include: Assembly, Hoisting, Transport, Erection, Launcher Standby, Oxidizer

Chilldown and Tanking, Fuel Chilldown and Tanking, Hold, Prelaunch Hold, Booster Engine Start, Launcher Rebound, Detank Fuel, Detank Oxidizer, Refilling, Launch, Transonic, Maximum αq Maximum g, Main Engine Starts, Main Engine Cut-offs, etc.

Centaur Flight Strain Gage Studies - Valuable data for use in determining fatigue spectra for prelaunch and flight conditions could be gleaned from the strain instrumented flight programs of earlier vehicles. Following is a brief summary of these studies.

Strain gages were installed on the liquid hydrogen tank at various locations near the forward and aft ends of the AC-3, -4 and -6 upper stage flight vehicles. The strain gage selection and circuitry on the AC-3 and -4 were of NASA design, while the AC-6 system was essentially a Convair design.

Preflight as well as flight data were planned as an aid in interpretation of results; however, the preflight information was not obtained during the AC-3 countdown and neither preflight nor flight data were analyzed by the computer for the AC-4 vehicle due to lack of funding and/or interest. Both preflight and flight data were analyzed for the AC-6 vehicle. The preflight data consisted of tank strains due to tank pressure changes of approximately 5 psi with: (1) insulation panels off at ambient temperature, (2) insulation panels on at ambient temperature, and (3) insulation panels on, tanked with liquid hydrogen. Thus the influences of pressure, temperature, and panel interference were recorded.

Reduction of the raw strain data was accomplished through the use of an IBM 7094 DCS digital computer and a SC 4020 plotter. Printed and/or plotted output included strains, stresses, temperatures, etc. For the AC-6 flight indicated bending moments and location of axis of bending were also evaluated. Results of the AC-3 flight were presented in Memo AS-D-954 (Ref. M-8). AC-4 data were recorded on tape but never analyzed by the computer program. AC-6 results were summarized in Memo AS-D-991 (Ref. M-9).

1.4.1.2.2 Fatigue Allowable Data - Structural Design Allowable Fatigue Strength Levels should be developed and utilized for the joint, material condition, and temperature ranges of interest (e.g., 301 EH at -423°F; 301 3/4 H, at -297°F; 301 1/2 H at -297°F and -423°F, etc.) The material/joint environmental test specification (0-71016, Ref. M-2) is of little value in establishing these allowables, since joints tested under it either pass or fail to pass the minimum "go", "no-go" limits. However, considerable test data do exist on the fatigue strength of base material, joints and small tanks as discussed in the section on the mechanical properties of materials and joints (Sections 1.3.1 and 1.3.2). These specimens of 301 stainless steel (of various hardnesses) were tested to failure in fatigue for various maximum stress levels and environmental conditions. The data could be supplemented with additional testing and checked (with the small tank data and supplementary analysis for uniaxial/biaxial effects) to make possible the determination of more realistic fatigue allowables. For each material condition, joint type, and temperature level, design allowable curves would be generated. The data points (stress level, S; log number of cycles to failure, log N and stress ratio, R) for each curve would be analyzed to determine the curve giving the best fit (in the least squares sense) and a complementary design allowable curve level would be established (e.g., see Figure 1).

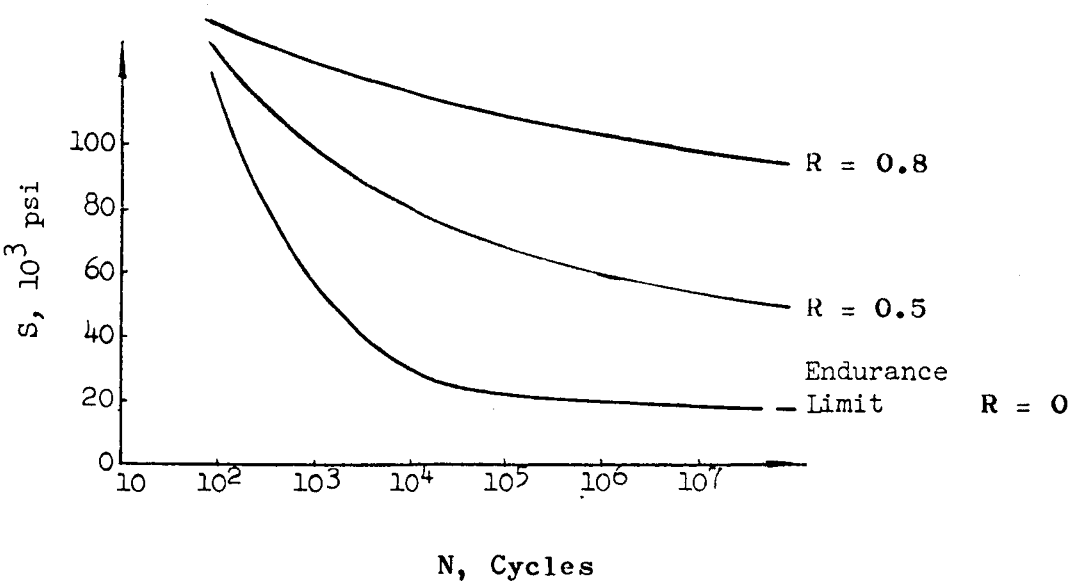

Figure 1 - Typical Fatigue Curve for Steel

1.4.1.2.3 Fatigue Analysis - A fatigue analysis of a uniform component uniaxially loaded from 0 to S_{max} to 0 for N cycles is quite simple. Introducing a steady state load on a new specimen to be loaded from S_{min} to S_{max} to S_{min} for N cycles is only slightly more complex. In more general situations, however, other techniques must be employed. For example, an aerospace vehicle pressure vessel will be subjected to various types of transients during its design life, e.g., normal operating transients caused by pressure, thrust, drag and gust changes, and emergency transients (i.e., thrust cut-off and launcher rebound). The normal operating transients may be expected to occur much more frequently than either start-up or emergency transients. This leads to the concept of cumulative fatigue damage, which considers each type of cyclic loading to "consume" a certain fraction of the design life, no one cycle causing fatigue failure by itself.

For example, during a component's lifetime it may be estimated that stress S_1 will be applied p_1 times; S_2, p_2 times; and S_n, p_n times. Each stress S_i will have a safe number of cycles, N_i, associated with it. For purely alternating loads these are given directly by a curve such as Figure 1. Then, a suitable criterion for satisfactory application, known as Miner's Hypothesis (Ref. M-10), is given by

$$\sum_{i=1}^{i=n} \frac{P_i}{W_i} \leq 1.0 \qquad \frac{P}{W} = \text{Damage ratio}$$

Tests have shown that if the lesser stresses are applied first, most material can endure a summation considerably in excess of 1.0; but, if the severe stresses are applied first, there is apparently some material damage which impairs its ability to carry the lesser loads, and the safe cumulative usage factor is likely to be less than 1.0.

Cumulative damage is assessed by plotting each stress grouping as shown in Figure 2 by point S_1'. The corresponding variable stress component for entering Figure 1 and determining N_1 is found as S_1 by the broken line construction shown in Figure 2.

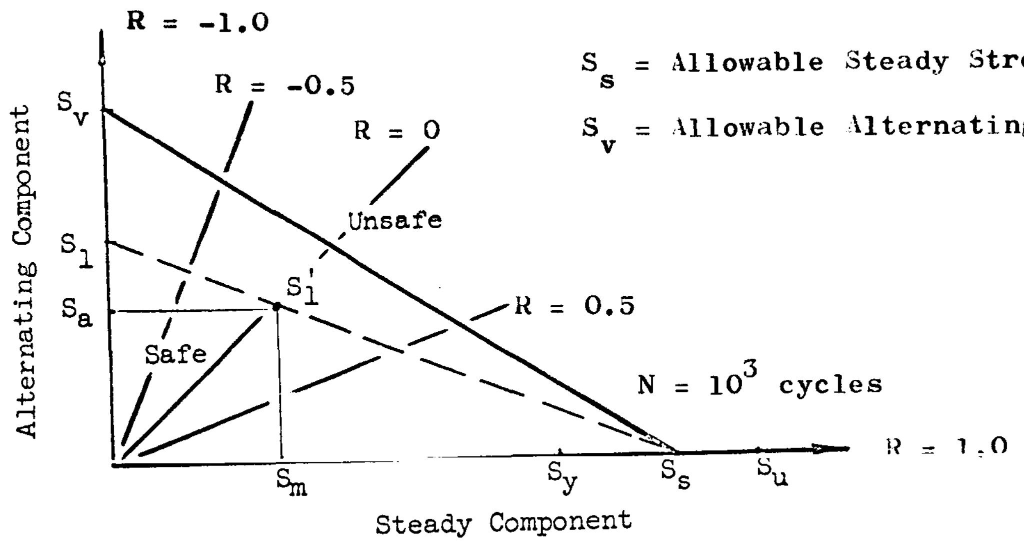

Figure 2 - The Modified Goodman Diagram

The presence of stress concentrators, e.g., corners or holes, complicates the fatigue analysis. Reasonably standard procedure (Ref. M-11, 12) include a stress concentration factor when calculating the variable stress component, but not for calculating the steady component. These stress components are then analyzed on the Goodman Diagram as discussed above. A more conservative approach is to apply the stress concentration factor to both variable and steady components. This second procedure, advocated by Langer, (Ref. M-13) has a more rational basis than the first.

Under certain circumstances, notably around stress concentrators (sharp corners, etc.) or because of non-uniform temperature distributions, the strain in certain localized areas may exceed the yield strain. The procedure suggested by Langer takes this local yielding into account and allows the analyst to calculate a new "mean stress" in certain cases. This aspect of fatigue analysis is given more consideration in the following paragraphs.

In aerospace vehicle applications the thermally loaded region is most often part of a plate or shell, and the loading is typically biaxial. A flat plate having a uniform temperature and clamped edges will develop elastic thermal stresses

$$\sigma_x = \sigma_y = -\frac{E\alpha T}{1-\nu} \ .$$

In order to clearly visualize the strains which do damage to the material let us suppose the thermal loading to occur in several stages. In Figure

3 square (a) is the original outline of the plate. If the plate is free and heated by a uniform temperature increment T, it "grows" to shape (b). From the previous equation, we know that the final stresses will be $\sigma_x = \sigma_y = -E\alpha T/(1-\nu)$ when the plate is returned to its original size, shape (a). Applying next only the x-stress, $\sigma_x = -E\alpha T/(1-\nu)$, the plate assumes shape (c). The total y-strain to be suppressed by applying $\sigma_y = -E\alpha T/(1-\nu)$ is thus

$$\epsilon_y^* = -\alpha T - \frac{\nu}{E}\frac{E\alpha T}{1-\nu} = -\frac{\alpha T}{1-\nu} .$$

The strains $\epsilon^* = -\alpha T/(1-\nu)$ are analogous to the outer fiber strains in the rotating beam specimen, and are a proper indication of damage done to the material during biaxial loading. We see that the elastic thermal stresses are calculated by multiplying ϵ^* by the modulus of elasticity, E.

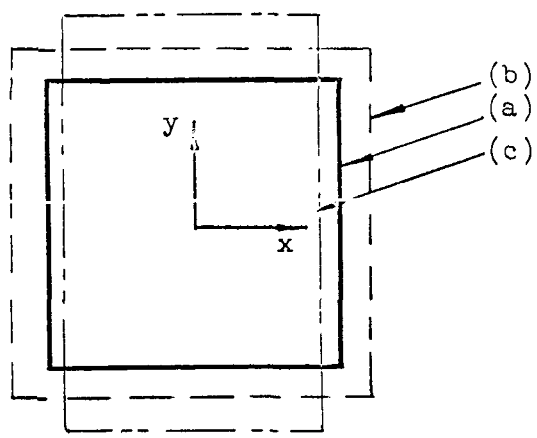

Figure 3 - Thermal Strains in a Plate

In the elastic range $\nu = 0.3$, but when the elastic limit is exceeded, $\nu = 0.3$ to 0.5 . Langer recommends using $\nu = 0.3$ for both elastic and plastic behavior, and absorbing any resulting error in the factor of safety. Thus, a fictitious elastic stress can always be defined which is related to the strain by $S' = E\,\varepsilon$.

In most structures, only a small volume of material is caused to yield by thermal expansions, e.g., a thin surface layer of a plate, and the gross behavior of the structure is dominated by the larger elastic regions. For this reason the strain pattern is little affected by localized yielding, and elastic solutions are appropriate for calculating strains. The small yielded region therefore has imposed upon it a certain cyclic strain pattern.

Consider now a hypothetical ductile material which yields at a constant value of Tresca equivalent stress, S_y , and is perfectly elastic at lesser values of equivalent stress. The stress-strain diagram is shown in Figure 4 (a) . So that the strain has a more obvious physical meaning we may visualize a simple bar in tension subjected to various cycles of controlled strain. We further assume, as is characteristic of ductile materials, that the material has equal yield strengths in tension and compression. It is clear that strain cycles with $|\varepsilon| \leq \varepsilon_y$ are elastic and are plotted on line OA , or its negative extension.

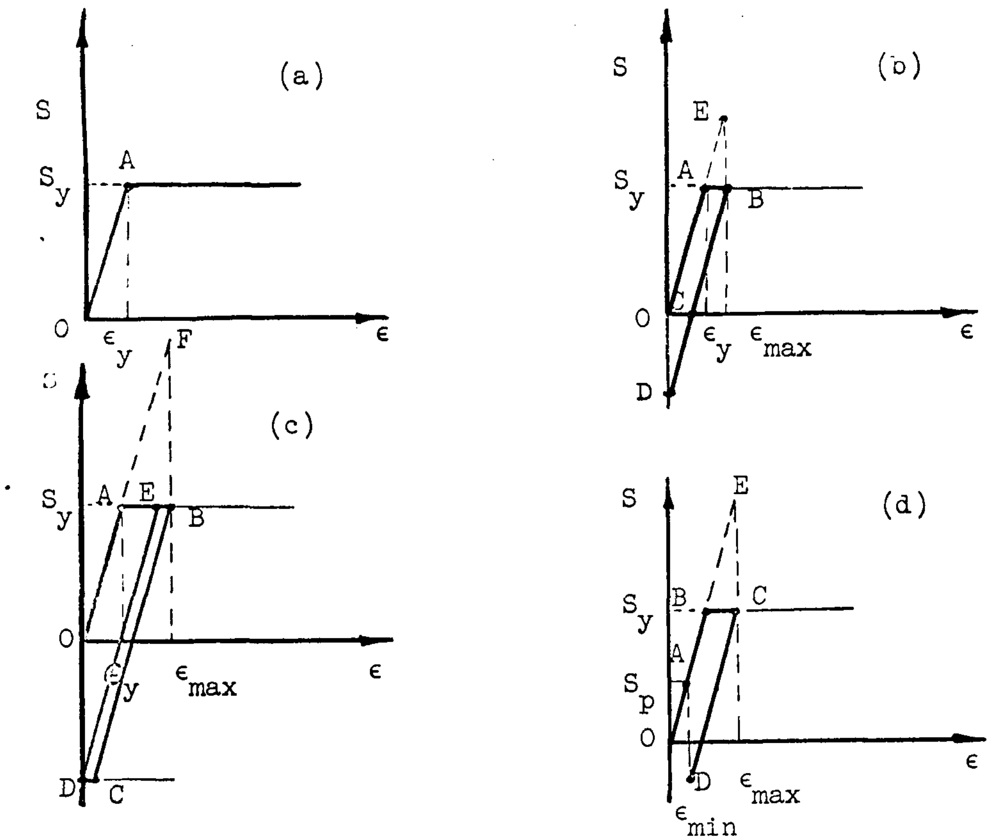

Figure 4 - Stress-Strain Diagrams for Various Strain Cycles of an Ideal Elastic-Plastic Material

Next consider a strain cycle with $0 \leq \varepsilon \leq \varepsilon_{max}$ such that $\varepsilon_y \leq \varepsilon_{max} \leq 2\varepsilon_y$ as shown in Figure 4 (b). When the strain reaches ε_y, the actual stress becomes limited to S_y until the strain begins to decrease from its maximum value at point B. The bar has now taken on a permanent plastic strain $\varepsilon_{max} - \varepsilon_y$ and thus unloads along a new elastic line BCD. At point C the stress has been reduced to zero, but the bar is longer than originally because of the plastic strain, and returning to $\varepsilon = 0$ at point D requires a compressive stress. The plastic yielding has had a beneficial effect, however, for when the strain cycle is repeated, behavior is completely elastic along path BCD. If, as earlier in this discussion, we calculate a fictitious elastic stress by $S' = E\,\varepsilon$, point E corresponds to the extreme limit of the first cycle. We may then calculate mean and alternating stress components by

$$S'_{mean} = \frac{S'_{max} + S'_{min}}{2} = \frac{S_e - 0}{2} = \frac{S_e}{2}$$

$$S'_{alt} = \frac{S'_{max} - S'_{min}}{2} = \frac{S_e - 0}{2} = \frac{S_e}{2}$$

From Figure 4 (b) we see that for subsequent cycles $S_{alt} = S'_{alt}$ has the same value, but S_{mean} is reduced to

$$S_{mean} = S'_{mean} - (S_e - S_y)$$

$$S_{mean} = S_y - S_e/2 \ .$$

The reduction in mean stress equals the amount by which the peak (fictitious elastic) stress exceeds the yield stress.

Now consider another case in which $\varepsilon_{max} > 2\varepsilon_y$, as shown in Figure 4 (c). The strain range is so great in this cycle that in unloading along path BC the material is forced to yield plastically in compression along path CD.

Subsequent cycles then occur along path BCDE, and there is plastic deformation at each extreme. Because the strain range is unaltered by this limiting phenomenon, our fictitious elastic stress, $S' = E\varepsilon$, has the same alternating component as during the initial cycle, i.e., $S_{alt} = S'/2$. The mean stress, however, has obviously been reduced to zero.

In the examples considered so far the minimum strain has been zero. Now suppose that a strain cycle with $\varepsilon_{max} - \varepsilon_{min} < 2\varepsilon_y$ is superimposed upon an initial stress, S_p, at point A in Figure 4 (d). The first strain application follows path ABC, but subsequent cycles follow the completely elastic path CD. The alternating stress component is not changed by this shift, but the mean stress is reduced to the same value that it would have taken if there had not been an initial stress. Langer cites several references to experimental work confirming this lack of influence of steady load on lifetime under plastic strain cycling.

If the material strain hardens appreciably, the above procedure may not be realistic. The increase in yield stress, S_y, which usually accompanies plastic flow lessens the reduction in mean stress illustrated in the above examples, and it is useful to define an elastic limit stress, S_b, for determining mean stress rather than using S_y. A reasonable approach which accounts for this effect is to let $S_b = S_y$ when S_y exceeds the endurance limit. For materials which strain harden appreciably the endurance limit may exceed S_y for the annealed material, and it is more realistic to take S_b as the endurance limit of polished specimens.

The ideas developed above may now be extended to more complexly loaded members by means of the equivalent stress concept. The Tresca equivalent stress is defined to be the numerical maximum of

$$S_{ij} = \sigma_i - \sigma_j$$

where σ_i and σ_j represent the principal stress components σ_1, σ_2, and σ_3. The following rules will enable S_{mean} and S_{alt} to be selected.

The alternating stress difference is given by

$$(S_{alt})_{ij} = \frac{(S'_{max})_{ij} - (S'_{min})_{ij}}{2}$$

for any given pair ij.

The mean stress difference will differ from

$$(S'_{mean})_{ij} = \frac{(S'_{max})_{ij} + (S'_{min})_{ij}}{\;}$$

according to the following rules.

(1) If $(S_{alt})_{ij} + (S'_{mean})_{ij} \leq S_b$, the mean stress difference is

$$(S_{mean})_{ij} = (S'_{mean})_{ij} .$$

(2) If $(S_{alt})_{ij} + (S'_{mean})_{ij} > S_b$, but $(S_{alt})_{ij} < S_b$, use

$$(S_{mean})_{ij} = S_b - (S_{alt})_{ij}$$

(3) If $(S_{alt})_{ij} \geq S_b$, then

$$(S_{mean})_{ij} = 0 \ .$$

Because fatigue failure occurs by the opening of small cracks in a tension field, it is reasonable to take $S_{mean} = 0$ whenever it is obviously compressive.

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